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CRANKSHAFT ASSEMBLY FOR INTERNAL COMBUSTION ENGINE

BACKGROUND OF THE INVENTION

Field of the Invention

5 The present invention relates to a crankshaft assembly including a flywheel, for an internal combustion engine. More specifically, the present invention relates to a crankshaft assembly for an internal combustion engine, which can effectively shift a resonance frequency of a flexural or bending vibration of the crankshaft assembly out of a target frequency band of a forced vibration which results such as during acceleration of a vehicle so as to effectively prevent occurrence of a thick sound or noise in an engine room, while ensuring a quick response of clutch engaging and disengaging operations, and/or which can prevent occurrence of a fore and aft vibration of a vehicle floor at the time of engagement of the clutch.

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Description of the Background Art

20 In a known crankshaft assembly for an internal combustion engine, a flywheel is directly connected to a crankshaft to use a mass of the flywheel mainly for reducing a torsional vibration generated in a rotating direction of the crankshaft assembly due to periodic torque fluctuation. However, the mass of the flywheel tends to generate a flexural or bending vibration in an axial direction of the crankshaft which causes a thick sound or noise in an engine room and thus in a vehicle compartment for an automotive vehicle, particularly at the time of the acceleration of the vehicle.

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35 Accordingly, there has been proposed a crankshaft assembly such as disclosed in Second Japanese Patent Publication No. 57-58542, wherein the flywheel is connected to the crankshaft through an elastic or flexible plate. The elastic plate has a rigidity in its rotating direction large enough for effectively

2

- 2 -

transmitting the power between the crankshaft and a transmission through a clutch, while the elastic plate has a rigidity in the axial direction small enough for shifting a resonance frequency of the bending vibration out of a frequency band of a forced vibration which results during the most frequently used engine speed (4,000 rpm) so as to overcome the above-noted problem.

However, the background art as mentioned above has the following problems.

When the rigidity of the elastic plate in the axial direction (hereinafter referred to as "the axial rigidity") is too small, a clutch stroke for engaging and disengaging the clutch is likely to become larger, resulting in a delayed response of the clutch engaging and disengaging operations leading particularly to failure of the clutch disengagement which is likely to cause such as an engine stall. On the other hand, when the axial rigidity of the elastic plate is too large, the deviation of the resonance frequency of the bending vibration from the target frequency band of the forced vibration can not be ensured.

Further, in the background art, when the flywheel is rotated, an axial run-out occurs on an engaging surface of the flywheel with a clutch facing of a clutch disc provided adjacent to the flywheel, due to a processing error and an assembling error of the elastic plate and the flywheel. Accordingly, when the clutch is engaged, a vibration is generated by a combination of the run-out of the engaging surface of the flywheel and the torque fluctuation of the engine, which is amplified by a vibration generated by the combustion in the engine cylinders and corresponding movements of associated members so as to cause a fore and aft vibration of the vehicle floor, which is uncomfortable for the driver and passengers in the vehicle compartment.

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- 3 -

SUMMARY OF THE INVENTION

Therefore, it is an object of the present invention to provide a crankshaft assembly for an internal combustion engine that can eliminate the above-noted defects inherent in the background art.

It is another object of the present invention to provide a crankshaft assembly for an internal combustion engine that can effectively shift a resonance frequency of a flexural or bending vibration of the crankshaft assembly out of a target frequency band of a forced vibration, particularly out of a target frequency band which results during acceleration of a vehicle so as to effectively prevent occurrence of a thick sound or noise in an engine room, while ensuring a quick response of the clutch engagement and disengagement operations so as to prevent particularly the failure of the clutch disengagement which is likely to cause such as an engine stall.

It is still another object of the present invention to provide a crankshaft assembly for an internal combustion engine that can prevent occurrence of a fore and aft vibration of a vehicle floor at the time of the engagement of the clutch by effectively eliminating an axial run-out of an engaging surface of a flywheel with a clutch facing generated during rotation of the flywheel.

To accomplish the above mentioned and other objects, according to one aspect of the present invention, a crankshaft assembly for an internal combustion engine comprises a crankshaft for transmitting a driving power to a transmission through a clutch, an elastic member fixed to the crankshaft, and a flywheel fixed to the elastic member such that the flywheel is supported in an elastic relationship with the crankshaft.

The flywheel has an engageable surface at a

- 4 -

side opposite to the elastic member in an axial direction of the crankshaft, and the engageable surface is engageable with an associated member of the clutch to receive a load therefrom in the axial direction when the
5 engageable surface is engaged with the associated member of the clutch.

The elastic member has a first predetermined rigidity in its rotating direction, the first predetermined rigidity being large enough to effectively
10 transmit the driving power to the transmission through the clutch. On the other hand, the elastic member has a second predetermined rigidity in the axial direction, the second predetermined rigidity being small enough to shift a resonance frequency of a bending vibration out
15 of a target frequency band of a forced vibration, while ensuring to prevent a failure of disengagement between the engageable surface of the flywheel and the associated member of the clutch.

According to another aspect of the present
20 invention,
a method for forming a crankshaft assembly for an internal combustion engine comprises steps of fixing a flywheel to an elastic member to form a unit, assembling the unit onto the crankshaft with the elastic member
25 mounted onto the crankshaft so as to support the flywheel in an elastic relationship with the crankshaft, and processing an engageable surface of the flywheel, which is engageable with an associated member of a clutch, based on an assembled condition between the
30 elastic member and the crankshaft so as to minimize an axial run-out of the engageable surface.

According to still another aspect of the present invention, a crankshaft assembly for an internal combustion engine comprises a crankshaft for
35 transmitting a driving power to a transmission through a clutch, an elastic member fixed to the crankshaft, and a

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- 5 -

flywheel fixed to the elastic member such that the flywheel is supported in an elastic relationship with the crankshaft.

5 The flywheel has an engageable surface at a side opposite to the elastic member in an axial direction of the crankshaft, and the engageable surface is engageable with an associated member of the clutch to control transmission of the driving power between the crankshaft and the transmission.

10 The engageable surface is designed to have an axial run-out which is no more than 0.1mm for ensuring a smooth engagement with the associated member of the clutch.

BRIEF DESCRIPTION OF THE DRAWINGS

15 The present invention will be understood more fully from the detailed description given hereinbelow and from the accompanying drawings of the preferred embodiment of the invention, which are given by way of example only, and are not intended to be limitative of
20 the present invention.

In the drawings:

25 Fig. 1 is a longitudinal cross section of a crankshaft assembly for an internal combustion engine according to a first preferred embodiment of the present invention;

a a Fig. 2 is a graph of ~~frequency versus~~ *versus frequency* vibration level, showing a shift of a resonance frequency of a flexural or bending vibration by changing a rigidity of an elastic or flexible plate in an axial
30 direction of a crankshaft;

Fig. 3 is a longitudinal cross section of a crankshaft assembly for an internal combustion engine according to a second preferred embodiment of the present invention; and

a 35 Fig. 4 is a graph of ~~run-out amount of~~ *fore and aft vibration of vehicle floor*
a ~~flywheel versus~~

- 6 -

no. a showing a relationship between an amount of an axial run-out of a flywheel and a fore and aft vibration of a vehicle floor.

DESCRIPTION OF THE PREFERRED EMBODIMENT

5 Now, a crankshaft assembly for an internal combustion engine according to preferred embodiments of the present invention will be described hereinbelow with reference to Figs. 1 to 4.

10 Fig. 1 shows a first preferred embodiment of the present invention. An engine crankshaft 1 is connected to pistons through respective connecting rods in a known manner for receiving the driving power therefrom. An elastic plate 2 substantially of a disc shape is fixed to one end of the crankshaft 1 by a
15 plurality of bolts 3. The elastic plate 2 is formed at its outer peripheral edge portion with an axially extending section 2a to which a ring gear R is fixed. The ring gear R engages with pinion gears of an engine starter motor for transmitting the driving power from
20 the engine starter motor to the crankshaft 1 when starting the engine.

An annular reinforcing member 4 is disposed between the elastic plate 2 and heads of the bolts 3. The reinforcing member 4 is formed at its outer
25 peripheral edge portion with a cylindrical section 4a extending in an axial direction of the crankshaft 1 and with a radially extending section 4b.

a A flywheel ^{*body*} 5 of an annular shape is fixed to the elastic plate 2 at their ^{*respective*} outer peripheral edge portions ^{*5a*} 5a and ^{*2b*} 2b through a plurality of bolts 6 and corresponding reinforcing members 7 disposed between the elastic plate 2 and heads of the bolts 6. The annular
a flywheel ^{*body*} 5 has a stepped inner peripheral edge surface defining a mounting opening 5b for receiving the
35 reinforcing member 4 therein. The stepped inner peripheral edge surface has a first section 5c extending

- 7 -

a 5 axially, a second section 5d extending radially outward from the first section 5c and a third section 5e extending axially from the second section 5d. The axial section 4a of the reinforcing member 4 is in a slidable contact with the first section 5c of the flywheel^{body} 5, and the radial section 4b of the reinforcing member 4 is spaced from the second section 5d of the flywheel^{body} 5 by a predetermined distance for allowing an axial movement of the flywheel along with the elastic plate 2. A radially extending inner surface 5f of the flywheel facing the elastic plate 2 is spaced apart from the elastic plate 2 by a predetermined distance for ensuring an elasticity of the elastic plate 2.

a 15 The flywheel^{body} 5 further includes a radially extending surface 5g at a side axially opposite to the radial surface 5f or the elastic plate 2. The radial surface 5g is engageable with a clutch facing 8 of a clutch disc 9 of a clutch in a known manner so as to control the transmission of the power between the crankshaft 1 and a transmission.

20 A rigidity of the elastic plate 2 in its rotating direction (hereinafter referred to as "the circumferential rigidity") is set large enough for effectively transmitting the power between the crankshaft 1 and the transmission through the clutch, while a rigidity of the elastic plate 2 in the axial direction (hereinafter referred to as "the axial rigidity") is set small enough for shifting a resonance frequency of the flexural or bending vibration out of a frequency band, of a forced vibration which results during the acceleration of the engine.

25 As described in the background art, when the axial rigidity of the elastic plate is too small, a clutch stroke for engaging and disengaging the clutch becomes larger, i.e. a clutch stroke loss gets larger, resulting in delayed response of the clutch engaging and

4

- 8 -

disengaging operations leading particularly to the failure of the clutch disengagement which is likely to cause such as an engine stall. On the other hand, when the axial rigidity of the elastic plate is too large, the deviation of the resonance frequency of the bending vibration from the target frequency band of the forced vibration can not be attained.

To overcome the above-noted problem, the axial rigidity of the elastic plate 2 in this embodiment is set to 600kg/mm to 2200kg/mm, wherein an axial displacement of the radial surface 5g of the flywheel 5 is no more than 1mm when an axial load or force 600kg to 2200kg is applied to the radial surface 5g. By selecting a value of the axial rigidity of the elastic plate 2 within the foregoing range, not only ^{is} the failure of the clutch disengagement ~~is~~ effectively prevented, but also the deviation of the resonance frequency of the bending vibration from the frequency band of the forced vibration, during the acceleration of the engine in this embodiment, is effectively attained so as to prevent generation of the thick sound or noise in the engine room.

Specifically, it is confirmed that the failure of the clutch disengagement, i.e. the failure of the disengagement between the radial surface 5g of the flywheel and the clutch facing 8 of the clutch disc 9, happens when an axial displacement of the radial surface 5g at the time of engagement with the clutch facing 8 exceeds 5% of a normal clutch stroke (normally at 7mm to 8mm) for engaging and disengaging the clutch. The normal clutch stroke is a distance between the radial surface 5g of the flywheel ^{body} 5 and the clutch facing 8 in a disengagement or released condition of the clutch. Accordingly, considering that an axial load applied to the flywheel ^{body} 5 through the clutch facing 8 is normally at 150kg to 200kg, the lower limit value 600kg/mm of the

- 9 -

axial rigidity of the elastic plate is selected, wherein the axial displacement of the radial surface 5g is within 5% of the normal clutch stroke when applied with the axial load 150kg to 200kg, as shown in TABLE 1.

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TABLE 1

	AXIAL LOAD	AXIAL RIGIDITY	AXIAL DISPLACEMENT
10	150 kg	500 kg/mm	0.30 mm (3.8 to 4.3 %)
	200 kg	500 kg/mm	0.40 mm (5.0 to 5.7 %)
	150 kg	600 kg/mm	0.25 mm (3.1 to 3.6 %)
	200 kg	600 kg/mm	0.33 mm (4.1 to 4.7 %)
15	150 kg	700 kg/mm	0.21 mm (2.6 to 3.0 %)
	200 kg	700 kg/mm	0.29 mm (3.6 to 4.1 %)

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(wherein, percentage denotes a rate of the axial displacement relative to the normal clutch stroke which is 7 to 8mm)

As seen from TABLE 1, the lower limit value 600kg/mm of the axial rigidity of the elastic plate 2 ensures the axial displacement of the radial surface 5g of the flywheel ^{body} 5 within 5% of the normal clutch stroke, i.e. the axial displacement of the radial surface 5g is between 0.25 to 0.33mm or between 3.1 to 4.7% relative to the normal clutch stroke when applied with the normal axial load at 150 to 200kg through the clutch facing 8, so that the failure of the clutch disengagement is effectively prevented. Naturally, the larger the axial rigidity of the elastic plate gets, the smaller the axial displacement of the flywheel gets.

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Now, the axial rigidity of the elastic plate 2 will be considered in view of shifting of the resonance

- 10 -

frequency of the bending vibration out of a frequency band of a forced vibration which results during the acceleration of the engine where the sound or noise generated by the bending vibration is the most significant. It is confirmed that the sound or noise generated by the bending vibration is effectively reduced when the resonance frequency is shifted out of the frequency band of the forced vibration during the acceleration of the engine.

a 10 Fig. 2 is a graph of ~~frequency versus~~ bending
a vibration level ^{versus frequency} showing a result of experiments using various elastic plates having different axial rigidities. The frequency band of the forced vibration during the acceleration of the engine is 200Hz to 500Hz.

15 In Fig. 2, a line A0 shows a relationship between the frequency and the bending vibration level without using the elastic plate, i.e. the flywheel is directly connected to the crankshaft. As can be seen, a resonance frequency of the line A0 is within 200Hz to

20 500Hz, which causes the sound or noise problem. A line A1 is derived by the elastic plate having the axial rigidity of 2200kg/mm, a line A2 is derived by the elastic plate having the axial rigidity of 1700kg/mm, a line A3 is derived by the elastic plate having the axial

25 rigidity of 1200kg/mm, and a line A4 is derived by the elastic plate having the axial rigidity of 1000kg/mm. As can be seen, the resonance frequency of each of the lines A1 to A4 is shifted out of the frequency band 200Hz to 500Hz, and further, the vibration level of each

30 of the lines A1 to A4 is considerably lower than the line A0 within the frequency band 200Hz to 500Hz. Though the line A1 has a vibration level higher than the line A0 around 200Hz, this happens in a very small range of frequency. Accordingly, the value 2200kg/mm is selected

35 as an upper limit value of the axial rigidity of the elastic plate, and the value 1700kg/mm is selected as a

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- 11 -

more preferable upper limit value of the axial rigidity.

In light of the above, the axial rigidity of the elastic plate 2 in this embodiment is selected at 600kg/mm to 2200kg/mm, and preferably at 600kg/mm to 1700kg/mm.

As understood from the above description, in this first embodiment, when the crankshaft 1 is rotated, the flywheel^{body} 5 is ensured to rotate with the crankshaft 1 by means of the large circumferential rigidity of the elastic plate 2. When the clutch is engaged and the engine is accelerated, the driving power is transmitted to the transmission with a very low bending vibration level by means of the axial rigidity of the elastic plate being no more than 2200kg/mm, so that the vehicle compartment can be kept quiet. On the other hand, when the clutch is disengaged, since the axial displacement of the flywheel is no more than 5% of the normal clutch stroke by means of the axial rigidity of the elastic plate being no less than 600kg/mm, the failure of the disengagement of the clutch is effectively prevented.

Fig. 3 shows a crankshaft assembly for an internal combustion engine according to a second embodiment of the present invention. In Fig. 3, the same or like parts or members are denoted by the same reference numerals. In the following description, explanations of those same or like members will be omitted to avoid redundant description. Further, though the clutch assembly is not shown in Fig. 3, the same clutch assembly including the clutch disc 9 and the clutch facing 8 is provided in the same manner as in Fig. 1.

In Fig. 3, the crankshaft 1 includes a stepped end surface having a first section 1a extending radially inward from its outer peripheral edge, a second section 1b extending axially from the inward end of the first section 1a toward the clutch disc 9, and a third

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- 12 -

circular section 1c extending radially from the second section 1b. The elastic plate 2 is of an annular shape having a mounting opening at its center for receiving the second section 1b therethrough. The elastic plate 2 is fixed to the crankshaft 1 with its axially extending inward end 2c facing the second section of the crankshaft 1 and with its radially extending inward end portion 2d facing the first section of the crankshaft. The other structure is substantially the same as in Fig. 1.

As mentioned in the background art, when the flywheel^{body} 5 is rotated through the crankshaft 1, an axial run-out is generated on the radial surface 5g due to the processing error and the assembling error of the elastic plate 2 and the flywheel^{body} 5 to cause the vibration when the clutch is engaged. The vibration further causes the fore and aft vibration of the vehicle floor.

In order to overcome the above-noted problem, in this embodiment, the radial surface 5g is processed in a manner to make an amount of the axial run-out no more than 0.1mm. Specifically, the processing of the radial surface 5g is performed in the following manner.

The flywheel^{body} 5 is first fixed to the elastic plate 2 by the bolts 6. Then, this unit is assembled to the crankshaft 1 with the axially extending inward end 2c of the elastic plate 2 facing the second section 1b of the crankshaft 1 and with the radially extending inward end portion 2d facing the first section 1a of the crankshaft. Then, the radial surface 5g is processed based on the assembled condition between the axially extending inward end 2c and the second section 1b and/or between the radially extending inward end portion 2d and the first section 1a to make the axial run-out of the radial surface 5g no more than 0.1mm.

By using the above-noted manner, the radial surface 5g is easily and precisely processed to make the

- 13 -

amount of the axial run-out no more than 0.1mm.

Fig. 4 is a graph of axial run-out amount of flywheel (radial surface 5g) versus fore and aft vibration of vehicle floor showing a result of experiments. It is confirmed that the fore and aft vibration of the vehicle floor which does not give a uncomfortable feeling to a human body is normally no more than 0.1G (gravitational acceleration). As can be seen from Fig. 4, a fore and aft vibration of the vehicle floor is substantially in direct proportion to an amount of the axial run-out of the radial surface 5g, and the fore and aft vibration becomes no more than 0.1G when the axial run-out becomes no more than 0.1mm. Accordingly, by making the amount of the axial run-out no more than 0.1mm as in this embodiment, the fore and aft vibration can be made no more than 0.1G.

As understood from the above description, in this second embodiment, when the crankshaft 1 is rotated, the flywheel^{body} 5 is ensured to rotate with the crankshaft 1 by means of the large circumferential rigidity of the elastic plate 2. Since the amount of the axial run-out of the radial surface 5g is no more than 0.1mm, the engagement between the radial surface 5g and the clutch facing 8 is performed quite smoothly, so that the fore and aft vibration does not exceed 0.1G. Accordingly, the driving power is transmitted from the engine to the transmission without giving the uncomfortable feeling to the human body.

It is to be appreciated that in this second embodiment, the axial rigidity of the elastic plate 2 is not necessarily selected at 600kg/mm to 2200kg/mm.

It is to be understood that the invention is not to be limited to the embodiments described above, and that various changes and modifications may be made without departing from the spirit and scope of the invention as defined in the appended claims.